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CONTROL SYSTEM FOR CONTINUOUSLY VARIABLE TRANSMISSION AND CONTINUOUSLY VARIABLE TRANSMISSION WHEREIN SUCH IS UTILISED

The invention relates to a control system for a continuously variable transmission or CVT and a CVT wherein such is utilised according to the preamble of claim 1.

Such control system and CVT are generally known, for example from the European patent publication EP-A-0.451.887. The CVT comprises a first rotatable pulley and a second rotatable pulley both provided with two pulley discs and with a piston/cylinder assembly for urging the pulley discs towards each other under the influence of a hydraulic cylinder pressure in said piston cylinder assembly. The pulley discs exert a clamping force on a drive belt, which is located between said pulley discs, such that torque transmission between said pulleys and said drive belt is enabled. The clamping force and the coefficient of friction between drive belt and pulley discs determine the maximum torque at which said torque transmission occurs virtually without mutual movement of drive belt and pulley discs, i.e. without belt slip. The ratio between the cylinder pressure in the piston/cylinder assembly of the first pulley and that in the piston/cylinder assembly of the second pulley determines the transmission ratio, i.e. the ratio between a rotational speed of the first and second pulley. The control system is capable of both clamping force control and transmission ratio control through determining the cylinder pressure for each of said pulleys. To this end the known control system is provided with a hydraulic circuit connecting the piston/cylinder assemblies to a pump and a reservoir for hydraulic medium and comprising two electronically controllable valves. The control system is furthermore provided an electronic control unit generating a respective control current for each valve in the hydraulic circuit at least based on a torque to be transmitted and on the rotational speed of the pulleys.

In the known control system the electronic control unit comprises two independently operating modules. A first module takes care of the clamping force control by setting the cylinder pressure in the piston/cylinder assembly of the first pulley through generating a first control current operating a first valve. Since the mechanical efficiency of the CVT decreases with increasing clamping force, the control system is usually arranged such that the cylinder pressure in the piston/cylinder assembly of the first pulley is maintained at a lowest possible level without belt slip occurring. Thereto the control system determines the minimum cylinder pressure in the first piston/cylinder assembly needed to prevent slip of the drive belt based on a torque to be transmitted by the transmission. In the known art said torque to be transmitted by the transmission is approximated on the safe side by adding to the actual torque level a value of 0.3 times

the maximum possible torque level. Thus, the safety factor with which the actual torque level is multiplied to calculate the torque to be transmitted by the transmission is thus 1.3 at the maximum possible torque to be transmitted, but increases rapidly with a decreasing level of the torque to be transmitted. The technical effect of applying a safety factor being that belt slip is prevented in a manner and at a level practically satisfactory, even during abrupt and/or unpredicted changes in the torque to be transmitted. A second module takes care of the transmission ratio control by setting the cylinder pressure in the piston/cylinder assembly of the second pulley through generating a second control current operating a second valve.

It is an object of the invention to provide for a solution to the desire to improve the performance and efficiency of the CVT by lowering the safety factor while still effectively preventing slip of drive belt.

Although, in general the known control system provides a simple and stable CVT control, it was found that after prolonged use, drive belts can show an unexpected amount of wear having a slight adverse effect on the durability of the CVT. It is a further object of the present invention to identify the problem underlying unexpected wear of the drive belt and to provide for a solution, thereby improving the durability of the CVT.

It was found that said wear is due to the occurrence of mutual movement of drive belt and pulley discs, i.e. belt slip, which suggested that during operation of the CVT situations can occur wherein the clamping force control is unable to adequately prevent belt slip. So, even though in the known art a relatively large safety factor is applied to the value for the torque to be transmitted, belt slip still remains a problem. These practical findings suggest that it is not possible to lower the safety factor without compromising the durability of the drive belt.

From US-A-5.707.314 a control system is known that is characterised by the feature that it determines a minimum cylinder pressure to prevent belt slip not only for the first pulley, but also for both for the first and the second pulley. In effect this means the clamping forces of both pulley are explicitly calculated. Based on these minimum cylinder pressures and on the ratio between the rational speeds of the first and second pulley, i.e. the transmission ratio, the control system then determines and controls both cylinder pressures to be equal, or higher than, the respective minimum cylinder pressure. With such system belt slip both on the first and second pulley may be prevent in most operational conditions.

However, according to the invention, it appeared that belt slip ~~only~~ occurred at a high rate of change of the transmission ratio. It was furthermore found that said belt slip occurs between the pulley discs of a specific pulley and more in particular, in case of the

CVT known from EP-A-0.451.887, the pulley discs of the second pulley, being the pulley used to control the transmission ratio.

According to the invention the control system and CVT having the features defined in the characterising portion of claim 1 advantageously prevents the occurrence of belt slip and in doing so provides for a CVT with an improved durability and efficiency. ~~The control system according to the invention is characterised by the feature that it determines a minimum cylinder pressure to prevent belt slip not only for the first pulley, but also for both for the first and the second pulley. In effect this means the clamping forces of both pulley are explicitly calculated. Based on these minimum cylinder pressures and on the ratio between the rational speeds of the first and second pulley, i.e. the transmission ratio, the control system then determines and controls both cylinder pressures to be equal, or higher than, the respective minimum cylinder pressure. In this manner it is realised that at all times the belt slip both between the pulley discs of the first and of the second pulley is effectively prevented. Although in most operational circumstances the minimum cylinder pressure in the first piston/cylinder assembly will be the most critical in respect to belt slip, the control system according to the invention will also prevent belt slip if the minimum cylinder pressure in the second piston/cylinder assembly is the most critical such as at a high rate of change of the transmission ratio.~~

It is noted that~~, in contrast to the state of the art where only one cylinder pressure is actively controlled,~~ the control system according to the invention determines both the cylinder pressure in the first and in the second piston cylinder assembly. It is common practice to use a feedback loop, wherein the actual value of a variable is fed back into a control system, when determining said variable. Therefore, it would appear that the control system according to the invention needs to be provided with two pressure sensors, one for each cylinder pressure. However, according to the invention, sufficient accuracy may be obtained, if only one cylinder pressure is measured using a pressure sensor and the other cylinder pressure is calculated from said measured cylinder pressure and the rotational speed of the pulleys as well as some transmission parameters. The cost of an additional pressure sensor thus being avoided.

Another advantage of the invention is that said approximation of the torque to be transmitted by the transmission may be made less safe compared to the known art without compromising the durability of the CVT. It was found that, when using the control system according to the invention, the torque to be transmitted by the transmission may be approximated by taking the level of the actual torque for each of said first and second pulley multiplied by a safety factor which is equal to 1.3 in a predominant part of a range of actual torque levels. The of specific size of said part depending on the transmission

ratio. In this manner a significant improvement of the overall efficiency of the transmission may be achieved.

According to ~~an elaboration of~~ the invention a cylinder pressure is maintained at a substantially higher level when a rate of change of the transmission ratio is relatively large than when said transmission ratio is constant. This may for example be effected by increasing the safety factor with an increasing rate of change of the transmission ratio. Since it was found that belt slip mainly occurs during a changing transmission ratio, the technical effect of this measure is that belt slip is prevented and that the performance and durability of the CVT is improved. According to the invention the relationship between the rate of change, RC, of the transmission ratio is given by the following equation:

$$\mathbf{RC} = -\mathbf{M}_{\mathbf{Nf/Ns}} \bullet \mathbf{Ns} \bullet \mathbf{Ff} \bullet \left(\mathbf{KsKf} - \frac{\mathbf{Fs}}{\mathbf{Ff}} \right)$$

wherein:

- $M_{Nf/Ns}$ is an experimentally determined (positive) parameter which varies with the ratio of the rotational speeds of said pulleys (4, 5),
- Ff is the force with which the drive belt (1) is clamped between the discs (8, 9) of the first pulley (4),
- Fs is the force with which the drive belt (1) is clamped between the discs (10, 11) of the second pulley (5),
- KsKf is the ratio of said forces Fs and Ff at which said rate of change would be zero.

From the equation it may be concluded that, if the transmission ratio defined as the rotational speed of the first pulley divided by the rotational speed of the second pulley increases, a desired rate of change of the transmission ratio may only be achieved at a certain minimum Fs and thus at a certain further minimum cylinder pressure in the piston/cylinder assembly of the second pulley. If the transmission ratio decreases, a desired rate of change of the transmission ratio may only be achieved at a certain minimum Ff and thus at a certain further minimum cylinder pressure in the piston/cylinder assembly of the first pulley. Thus, if the control system of the CVT is provided with an input signal representing a desired rate of change of the transmission ratio, a further minimum cylinder pressure may be determined either for the first or for the second pulley using the said equation. Based on this further minimum cylinder pressure, on said minimum cylinder pressures and on the transmission ratio, the control system may then determine and control both cylinder pressures to be equal to, or higher than, the respective minimum and/or further minimum cylinder pressures. In this manner

it is at all times possible to effectively prevent belt slip both between the pulley discs of the first and of the second pulley and to reliably achieve a desired rate of change of the transmission ratio.

According to a further development of the invention, the control system is provided with change rate restriction means that are capable restricting the rate of change of the transmission ratio in case the flow of hydraulic medium delivered by the pump is insufficient to maintain the desired cylinder pressures. Said change rate restriction means may be able to determine the difference between the actual flow of hydraulic medium delivered by the pump and the flow required to achieve said further minimum

cylinder pressure and thus to achieve said desired rate of change. Said change rate restriction means are then arranged such that, as long as said difference is negative, the desired rate of change is reduced. Alternatively, the change rate restriction means may be able to determine whether the hydraulic system is capable of controlling an actual cylinder pressures to a desired level. If the change rate restriction means detect that the actual cylinder pressure does not reach the desired level, the desired rate of change is reduced. The change rate restriction means ensure that the hydraulic system is at all times capable of generating said further minimum cylinder pressure and belt slip is prevented even when the originally desired rate of change of the transmission ratio would yield belt slip.

As is known in the art, the response of the hydraulic system and of rotating parts of the transmission to a change in the control currents generated by the electronic control unit is often non-linear. Because of said non-linear response the control system is usually unable to accurately control the cylinder pressures to their respective desired level. This non-linear behaviour has an adverse effect on the accuracy with which the cylinder pressures may be controlled and therewith on the efficiency of the transmission. It is, for instance, not feasible to use the integrating action of a PID-regulator due to said non-linear behaviour. Fast changes of the transmission ratio are difficult or even impossible to achieve reliably, because for in this case relatively large differences between the cylinder pressure of the first and of the second pulley are needed, in which case the non-linear behaviour is even more pronounced. This is especially important if a relatively small safety factor is adopted, as is allowed by control system according to the invention. It is therefore advantageous for the control system to be able to account for the non-linear response of the transmission. To this end the control system may be provided with linearisation module containing a mathematical representation of a mass balance of the hydraulic system. Such a mass balance may be construed in a known manner from the flow of hydraulic medium circulated by the pump, the flow characteristic of the valves, and the flow to and from the piston/cylinder assemblies when the transmission ratio changes. Incorporating leakage flows and a value for the compressibility of the hydraulic medium in said mass balance can improve the accuracy of the control system even further. The mathematical representation of the mass balance is incorporated in the linearisation module of the control system such that the system can determine one or more control current required for effecting a desired pressure response in the hydraulic circuit. In a further development of the invention the mathematical representation is contained in a sub-unit which is physically separable

from the electronic control unit. This allows the control system to be easily adopted to several layouts of the hydraulic system by exchanging the sub-unit.

According to the invention it is preferable to use a mathematical representation of the hydraulic circuit based on flows of hydraulic medium, because the said flows may be derived directly from known transmission variables such as the rotational speed of a pulley drivingly associated with the pump and the rate of change of the transmission ratio, which determines the flows to and from the piston/cylinder assemblies of the pulleys.

According to a further elaboration of the invention the electronic control unit may comprise separate control modules for clamping force control and for transmission ratio control. The first control module controls the clamping force and is capable of generating a first control current for operating the first valve and the second control module controls the transmission ratio and is capable of generating a second control current for operating the second valve. This set-up of the electronic control unit is particularly suited for a hydraulic system wherein the first valve is a pressure control valve which determines the cylinder pressure of the first pulley and the second valve is a flow control valve which controls a flow of hydraulic medium to and from the piston/cylinder assembly of the second pulley. This type of hydraulic system allows the cylinder pressure of the second pulley to become only as high as the cylinder pressure of the first pulley. However, the design of the piston/cylinder assemblies is such that clamping force of the first pulley can become higher than that of the second pulley. The second control module incorporates both a PI-regulator and said linearisation module, so that the control of the cylinder pressure of the second pulley is more accurate and the transmission ratio is controllable with a high degree of accuracy. The first control module determines said minimum cylinder pressures and said further minimum cylinder pressure, as well as a desired ratio of the first and second cylinder pressures, based on torque signal representing a torque to be transmitted by the transmission multiplied by a safety factor, signals representing the rotational speed of the pulleys and a signal representing a desired rate of change of the transmission ratio. Subsequently, the first control module selects the cylinder pressure for the first pulley to be the highest of a cylinder pressure:

- given by the minimum cylinder pressure in the piston/cylinder assembly of the first pulley, or

- given by said desired ratio of the first and second cylinder pressures and a minimum cylinder pressure in the piston/cylinder assembly of the second pulley, or

- given by the further minimum cylinder pressure in the piston/cylinder assembly of the first pulley, or

- given by said desired ratio of the first and second cylinder pressures and a further minimum cylinder pressure in the piston/cylinder assembly of the second pulley.

Based on a pressure difference signal representing the difference between said highest cylinder pressure of the first pulley and the actual cylinder pressure of the first pulley measured by means of a pressure sensor and on the valve pressure/current characteristic, an appropriate control current for operating the first valve is generated by the first module. The second control module determines the transmission ratio by controlling the rotational speed of the second pulley based on a speed difference signal representing the difference between a measured rotational speed of the second pulley and a desired rotational speed of the second pulley which is calculated from the measured rotational speed of the first pulley and the desired transmission ratio. The first and second control modules are capable of mutually providing each other with a signal at least representing the cylinder pressure in the first piston cylinder assembly and said desired rotational speed of the second pulley. This elaboration of the invention has the advantage, that the control system may be used in combination with the so called Master/Slave hydraulic layout which is widely used in contemporary CVTs and provides for a relatively simple and cost effective transmission control.

According to the invention the CVT is suited for application in a motor vehicle having an engine with an engine shaft and a drive shaft for driving driven wheels of the vehicle, in which case it is preferred that the first pulley is drivingly connected to the first pulley and the engine shaft is drivingly connected to the second pulley. The control system and CVT according to the invention are particularly suited for use in combination with an adjustable clutch applied in the torque path between engine and driven wheels, which clutch is adjusted to be capable of transmitting a maximum amount of torque, said maximum amount of torque being smaller than a torque transmittable by the continuously variable transmission without relative movement between the drive belt and the pulley discs. The technical effect of such a combination of adjustable clutch and CVT according to the invention being that belt slip virtually eliminated and that a small safety factor may be applied improving the fuel efficiency of the motor vehicle.

The invention will now be elucidated further with reference to the figures.

Figure 1 shows a mechanical layout of a CVT according to the prior art.

Figure 2 shows a electro-hydraulic control system according to the prior art.

Figures 3 through 5 are schematic representations of control systems according to the invention.

Figure 6 shows a graph of the safety factor in relation to the actual torque.

Figure 7 shows a CVT in a vehicular drive line.

In figure 1 a schematic representation of a CVT is shown. The CVT comprises a first rotatable pulley 4 and a second rotatable pulley 5 both provided with two pulley discs 8, 9 and 10, 11 respectively as well as a piston/cylinder assembly 12, 13 respectively for urging the pulley discs 8, 9, 10 and 11 towards each other under the influence of a hydraulic cylinder pressure Pf and Ps respectively in said piston cylinder assembly 12, 13 respectively. The pulley discs 8, 9, 10 and 11 exert a clamping force on a drive belt 1, which is located between said pulley discs 8, 9, 10 and 11, such that torque transmission between said pulleys 4 and 5 and said drive belt is enabled. The cylinder pressures Pf and Ps are maintained by allowing hydraulic medium to and from the respective piston cylinder assembly 12 and 13 via channels 6 and 7. The clamping force and the coefficient of friction between the pulley discs 8, 9, 10, and 11 and the drive belt 1 determine the maximum torque at which said torque transmission occurs virtually without mutual movement of drive belt 1 and pulley discs 8, 9, 10 and 11. The ratio between the cylinder pressure Pf and Ps determines the transmission ratio, i.e. the ratio between a rotational speed of said pulleys 4 and 5. The CVT is thus capable of transmitting an amount of torque from the first pulley 4 to the second pulley 5 at a transmission ratio which is continuously variable within a range of possible transmission ratios. Such CVTs are widely used in particular in automotive applications.

The electro-hydraulic control system 15, 16, 17, 18, 20, 21 shown in figure 2 is generally known as the Master/Slave hydraulic layout, since at any given time the cylinder pressure Ps in the piston/cylinder assembly 13 of the second or slave pulley 5 is bound by the level of the cylinder pressure Pf in the piston/cylinder assembly 12 of the first or Master pulley 4. The piston/cylinder assembly 13 of the second pulley 5 has a larger piston surface area than that of the first pulley 4, so that, although the pressure ratio Ps/Pf is 1 at most, the ratio of the clamping force exerted by the first pulley and the clamping force of the second pulley 5 may become larger than 1. Usually the ratio between the piston surfaces of the second pulley 5 and the first pulley 4 is about 2 to 2.5. The main functions of the control system 15, 16, 17, 18, 20, 21 are to, on the one hand, control the cylinder pressures Pf and Ps in the first and second piston/cylinder assemblies 12, 13, so that a desired speed ratio of the rotational speeds of the pulleys 4 and 5 is maintained, i.e. ratio control, and to, on the other hand, ensure that torque can be transmitted between the drive belt 1 and the first and second pulleys 4 and 5 virtually without mutual relative movement, i.e. clamping force control.

The control system 15, 16, 17, 18, 20, 21 is provided with a hydraulic circuit 15, 16, 20, 21 having two valves 15 and 16 as well as with an electronic control unit 17, 18, 19 comprising two control modules for generating control currents If and Is to operate said valves 15 and 16. A Pump 20 is present for generating a flow of hydraulic medium from a reservoir 21. The valves 15 and 16 determine the cylinder pressure Pf and Ps in the first and in the second piston/cylinder assembly 12 and 13 respectively by allowing hydraulic medium from the pump 20 to said assemblies 12 and 13. Excess medium in the hydraulic system is discharged into said reservoir 21. In the known art, the electronic control unit 17, 18 has a first control module 17 which performs the clamping force control and a second control module 18 which performs the ratio control. The clamping force control and the ratio control are completely independent, so that a simple and adjustable control system 15, 16, 17, 18, 20, 21. Such an approach, however, has the disadvantage mentioned in the above.

Figures 3 through 5 show schematic representations of the control system 15, 16, 17, 18, 19, 20, 21 according to the invention. It is noted that these specific representations are intended only for clarification of the invention. In practice the control system 15, 16, 17, 18, 19, 20, 21 according to the invention may be effected in several ways.

In figure 3 a graphical representation of the main function blocks I, II, III and IV which may be recognised in the control system 15, 16, 17, 18, 19, 20, 21 according to claim 1 is shown. Block I has inputs for three signals, being a signal Nf representing the rotational speed of the first pulley 4, a signal Ns representing the rotational speed of the second pulley 5 and a signal Tt representing the torque to be transmitted by the transmission. From these three signals a minimum first cylinder pressures Pf,min and a minimum second cylinder pressure Ps,min required for torque transmission between the drive belt 1 and the pulleys 4 and 5 substantially without mutual relative movement are determined. Constants such as a coefficient of friction of the contact between pulley disc 8, 9, 10 or 11 and the drive belt 1 and a surface areas of pistons of the piston/cylinder assemblies 12, 13 are taken into account in a manner known to the man skilled in the art. Subsequently, in block II the cylinder pressures Pf and Ps are selected at least such that they are larger than the minimum required cylinder pressures Pf,min and Ps,min respectively. Finally, in block III, the control modules generates the appropriate control currents If and Is for controlling said valves so that the cylinder pressure of the first pulley 4 is equal to Pf and that the cylinder pressure of the second pulley 5 is equal to Ps. The selection of Pf and Ps made in block II usually also has to satisfy the requirement that a certain transmission ratio is obtained or maintained. To this end block

IV may be present. In block IV the cylinder pressure ratio Pf/Ps of the first cylinder pressure Pf and of the second cylinder pressure Ps is determined based on said signals Nf, Ns and Tt as well as a signal Nf/Ns representing the ratio of the rotational speeds of said pulleys 4 and 5. In block II the pressures Pf and Ps may now be selected by making either Pf equal to Pf,min or Ps equal to Ps,min and by selecting the other cylinder pressure Ps or Pf respectively so that said required cylinder pressure ratio Pf/Ps is obtained. For efficiency reasons it is desirable to select the lowest possible cylinder pressures Pf and Ps, so that it may be concluded that, if the ratio of the cylinder pressure Pf of the first pulley 4 and of the cylinder pressure Ps of the second pulley 5 is larger than 1 Ps is made equal to Ps,min and if the latter ratio is smaller than 1 Pf is made equal to Pf,min.

In figure 4 a graphical representation of the main functional blocks I, II, III, IV and V which may be recognised in the control system 15, 16, 17, 18, 19, 20, 21 according to claim 4 is shown. In figure 4 function block V has been added to the function blocks I, II, III and IV shown in figure 3. Block V has a signal ROC representing a desired rate of change for the transmission ratio as an input. As was explained in the above, in order to achieve a certain rate of change of the transmission ratio a further minimum cylinder pressure Pff,min, Psf,min respectively can be determined for cylinder pressure Ps, Pf respectively in the piston/cylinder assembly 12, 13 respectively of the pulley 4 or 5 which rotational speed decreases during a change of the transmission ratio. Subsequently, in block II the cylinder pressures Pf and Ps are selected at least such that they are larger than both the minimum required cylinder pressures Pf,min, Ps,min respectively and the further minimum Pff,min or Psf,min. Again, the cylinder pressure Pf and Ps must also satisfy a cylinder pressure ratio Pf/Ps as prescribed by block IV.

Figure 5 shows a graphical representation of the electronic control unit 17, 18, 19 of a control system 15, 16, 17, 18, 19, 20, 21 particularly suited for use in combination with the so called Master/Slave hydraulic layout. The electronic control unit 17 ,18, 19 comprises separate control modules 17 and 18 for clamping force control and for transmission ratio control. The first control module 17 controls the clamping force and is capable of generating a first control current If for operating the first valve 15 of the Master/Slave hydraulic system and the second control module 18 controls the transmission ratio and is capable of generating a second control current Is for operating the second valve 16. The second control module 18 incorporates both a PI-regulator having a linear response and a linearisation module 19, so that the control of the cylinder pressure Ps of the second pulley 5 is more accurate and the transmission ratio is controllable with a high degree of accuracy. The first control module 17 determines

the minimum cylinder pressures Pf,min and Ps,min and said further minimum cylinder pressure Pff,min or Psf,min, as well as a ratio of the first and second cylinder pressures Pf/Ps, based on torque signal Tt representing a torque to be transmitted by the transmission Tp multiplied by a safety factor Sf, signals Nf and Ns, respectively representing the rotational speed of the pulleys, and a signal ROC, representing a desired rate of change of the transmission ratio. Subsequently, the first control module 17 selects the cylinder pressure Pf for the first pulley 4 to be the highest HPf,min of a cylinder pressure:

- given by the minimum cylinder pressure Pf,min in the piston/cylinder assembly 12 of the first pulley 4, or
- given by said ratio of the first and second cylinder pressures Pf/Ps and a minimum cylinder pressure Ps,min in the piston/cylinder assembly 13 of the second pulley 5, or
- given by the further minimum cylinder pressure Pff,min in the piston/cylinder assembly 12 of the first pulley 4, or
- given by said ratio of the first and second cylinder pressures Pf/Ps and a further minimum cylinder pressure Psf,min in the piston/cylinder assembly 13 of the second pulley 5.

Based on a pressure difference signal representing the difference between said highest cylinder pressure HPf,min of the first pulley 4 and the actual cylinder pressure Pf,a of the first pulley 4, measured by means of a pressure sensor, and on the pressure/current characteristic of the first valve 15, an appropriate control current If for operating the first valve 15 is generated by the first module 17.

The second control module 18 determines the transmission ratio by controlling the rotational speed of the second pulley based on a speed difference signal representing the difference between a measured rotational speed Ns of the second pulley 5 and a desired rotational speed Ns,d of the second pulley 5 which is calculated from the measured rotational speed Nf of the first pulley 4 and the desired transmission ratio Nf/Ns,d. The first and second control modules 17 and 18 are capable of mutually providing each other with a signal Sf and Ss at least representing the cylinder pressure Pf in the first piston cylinder assembly 12 and said desired rotational speed Ns,d of the second pulley 5.

Figure 6 is a graph illustrating the dependency of the torque level represented by a torque signal Tt that is equal to an actual torque to be transmitted Tp multiplied by a safety factor Sf. The line T=Tp represents the actual torque to be transmitted by the transmission. The line marked 'Tt, prior art' represents the torque level represented by the signal Tt according to the prior art. After dividing Tt by Tp the dotted line is found that

represents a safety factor Sf according to the prior art. It can be seen that only at the maximum torque Tp,max said safety factor Sf is equal to 1.3.

According to the invention the safety factor Sf is constant and equal to 1.3 in a large part of the range of possible torque values from Tp,min through to Tp,max as is illustrated in figure 6. The torque level represented by the signal Tt is thus found by multiplying Tp with 1.3. At torque values below Tg the safety factor Sf according to the invention is made to increases rapidly, so that the product of Sf and Tp remains essentially constant at 1.3 times Tg. The latter is necessary, because inevitable disturbances in the actual torque to be transmitted Tp have an absolute character, so that their influence is particularly significant in the lower part of said range of possible torque values. The strategy for the level of the safety factor according to the invention allows for a relatively simple determination of the torque signal Tt. From figure 6 it appears that on average the safety factor Sf according to the invention is smaller than the safety factor according to the prior art. This effect of this being that on average the clamping forces are smaller and the CVT efficiency is improved as a consequence.

Figure 7 shows a schematic representation of the application of a CVT in the drive line of a motor vehicle. The motor vehicle is provided with an engine 22 having an engine shaft 23. The engine shaft 23 being drivingly connected to the second pulley 5 of a CVT. The first pulley 4 of the CVT is drivingly connected to a drive shaft 24 for driving the driven wheels 25 of the motor vehicle. During operation of the motor vehicle driving power is transmitted from the engine 22 through the drive belt 1 to the driving wheels, whereby the CVT is capable continuously varying the ratio of drive shaft torque and drive shaft rotational speed. An adjustable clutch 26 may be adopted in the drive line at a location between the engine 22 and driven wheels 25. If the clutch is adjusted to be able to transmit an amount of torque which is somewhat smaller than the torque to be transmitted by the transmission Tp, the occurrence of belt slip due to unexpected changes in a drive line torque level is effectively avoided. Such unexpected changes may originate from the driven wheels 25, for example when they run over a bump or a hole in a road surface.